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CONSTRUCTION AND DESIGN OF EQUIPMENT FOR PUNCHING HOLES BY EXPLOSION

KONSTRUKTSIYA I RASCHET OBORUDOVANIYA DLYA VZRYVNOI
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Construction and design of equipment for
punching holes by explosion

V.G. Kononenko, K.I. Zaitsev

At the present time a new method of punching holes in thick plates and sections is more and more widely employed in factory workshops and machine shops – namely the punching of holesby explosion (1,2).

The plant for effecting the operation consists of the following basic components: cartridge chamber 1 (fig.1), barrel 2, silencer 3, the working parts – punch 4 and die 5 and the frame 6 which ensures the coaxiality of the punch and die and takes up the load during the punching process.

In utilising the energy of the explosion the punch acquires a considerable velocity when approaching the part (up to 100 m/sec) and it was therefore possible to design a unit weighing only 25 kg, but developing very great power. The work done by the unit can be equivalent to the work done by a 50 to 75 ton press.

Fig.1   Diagrammatic view of a unit for punching holes
by explosion

The plant for punching holes by explosion can be correctly designed only if the laws governing the behaviour of the metal when subjected to high speed loads and the technological features of the process are known. These were ascertained by studying the process of punching holes in high speed impact machine tests (3).

In the course of these experiments it was found that alongside with good technological indices (high accuracy and quality of the operation, satisfactory stability of the tool, absence of microfissures), rapid punching of holes also shows satisfactory indications as far as the work done and energy distribution are concerned. The use of explosion devices for punching holes makes it possible to obtain holes of considerable diameter (up to 25 mm) in thick plates (up to 15 to 18 mm) with a weight of powder equal to approximately 2g.

The initial data for designing the unit are the characteristics of the holes (diameter, mark, specification and thickness of the material) to be punched, and also the coordinates of the holes in the detail or part. The latter are necessary for deciding the dimensions of the frame bracket (overhang and jaw).

Knowing the characteristics of the hole to be punched, we can determine the work done in the deformation process from the following relationship

$$A_\partial = K_v A_{stat} \tag{1}$$

where A_∂ is the required work done in the deformation process when using the explosion method, K_v is the dynamic coefficient (depending on the approach velocity of the punch to the part and on the properties of the material being punched) and A_{stat} is the work done in the static deformation of the hole.

For structural steels the dynamic coefficient can be taken from the graph shown in fig.2, which is based on experimental data. Theoretical values of dynamic coefficients, although coinciding with the experimental ones, cannot be recommended for practical use owing to the laborious calculations involved in obtaining them.

Fig.2 <u>Relationship of the dynamic coefficient to the impact velocity of the punch</u>

It will be seen from fig.2 that the value of the dynamic coefficient is less than one at high deformation velocities i.e. at considerable velocities the work done in dynamic deformation (explosive punching) is less than the work done in static deformation (usual punching in a press). This is clearly illustrated by the curves shown in fig.3.

<u>Fig.3</u> <u>Graph showing the relationship of the work done in deformation to the thickness of the material and the impact velocity</u>

_____ explosion punching

— — — — — — — press punching

The change in the deformation work can be explained by the simultaneous effect of two mutually counteracting factors. An increase in the deformation velocity brings about an increase in the resistance to deformation due to the effect of inertia forces, which increases the total work done. On the other hand the increased temperature at the seat of the deformation and the decreased volume of material subjected to plastic deformation, decreases the total work done in deformation. At very high deformation velocities ($v_o >$ 35 to 40 m/sec) the second factor predominates.

The work done in static deformation A_{stat} in the usual press process is obtained (4) from the expression

$$A_{stat} = \frac{\lambda \pi d_n S^2 \tau_o}{1000} \quad kgm \qquad (2)$$

where λ is a coefficient representing the ratio of the mean stress to the maximum stress (for punching holes in thick-gauge steel parts λ = 0.45 to 0.30), d_n is the diameter of the punch in mm, S is the thickness of the material being punched in mm and τ_o is the resistance to shear in kg/mm^2 .

For deformation velocities different to those shown in fig.2 or for punching holes in other metals (i.e. if the value of the dynamic coefficient is not known), conditional use can be made of a velocity coefficient α proposed by S.I. Gubkin (5) and equal to 1.05 to 1.15. In this case the result will always be on the high side giving an increased value for the deformation work i.e. the design will at first tend to an over-estimate of the energy carrying charge (the powder), which can be subsequently adjusted by trial tests.

The basis of the design are the ballistic and strength calculations. The ballistic calculations concern the cartridge chamber, the barrel and the punch and the calculations for strength concern all the stressed elements of the unit (frame, barrel and others).

The aim of the ballistic design is to choose such parameters for the unit (diameter and length of barrel, dimensions of cartridge chamber, weight of powder charge and weight of plunger), that the calculated impact velocity of the punch against the part is an optimum velocity. The optimum impact velocity for a given weight of the plunger must be taken as the velocity corresponding to the kinetic energy of the plunger equal to the deformation work A_∂. This velocity can be found from the relationship

$$A_\partial = \frac{m_n v_o^2}{2} \qquad (3)$$

where m_n is the mass of the plunger and v_o is the optimum velocity of the plunger at impact with the part.

The design velocities of the unit can be approximately arrived at by the following method.

Since the firing process constitutes an adiabatic process of expansion of gases formed as the result of the combustion of the powder, we can write (6):

$$W = \frac{1}{n-1} \, p_1 V_1 \left[1 - \left(\frac{V_1}{V_2} \right)^{n-1} \right] \qquad (4)$$

where W is the work done in the expansion of the gases, p_1 is the initial pressure of the gases, V_1 and V_2 are the volumes of the gases at the beginning and at the end of the expansion process (V_2 is determined if the diameter and the length of the barrel are known) and $n=1.2$ is the polytropic index.

On the basis of experiments using cartridges of 7.62 caliber and cut-off rifle cartridges of 32 caliber, the values of V_1 can be taken as equal to 0.0048 and 0.0045 cubic decimetres respectively.

The initial pressure of the powder gases is determined (7) from the following internal ballistics equation:

$$p_1 = \frac{f \Lambda}{1 - \alpha \Delta} \quad kg/dm^2 \qquad (5)$$

where is the density of the charge

$$\Delta = \frac{\omega}{V_1} \quad kg/dm^3 \qquad (6)$$

ω is the weight of the powder charge, f is the force of the powder and α is the co-volume of the powder.

The powder characteristics can be taken from the following table:

Powder	f in kg. dm/kg	α in dm^3/kg.
Pyroxilin (nitrocellulose)	$(7.7-9).10^5$	0,9-1.1
Nitroglycerin	$(9-11).10^5$	0.75-0.85
Black powder	$(2.8-3).10^5$	0.5

Substituting the value p_1 from equation (5) in equation (4), we get the work done in the expansion of the gases:

$$W = \frac{1}{n-1} \cdot \frac{f \Delta}{1 - \alpha \Delta} \left[1 - \left(\frac{V_1}{V_2} \right)^{n-1} \right] V_1 \qquad (7)$$

Substituting in equation (3) (on the basis of conditions prevailing during the process of punching the hole) the value of the calculated design velocity v_p instead of the optimum velocity, we can determine the work done by the punch

in the process of deformation.   Clearly this work is equal to the work done by the expanding gases and we can therefore say:

$$\frac{m_n v_p^2}{2} = \frac{1}{n-1} \cdot \frac{f\Delta}{1-\Delta\alpha} \left[1 - \left(\frac{V_1}{V_2}\right)^{n-1} \right] V_1$$

whence the calculated velocity of the approach of the punch is

$$v_p^2 = \frac{2}{n-1} \cdot \frac{f\Delta}{1-\alpha\Delta} \left[1 - \left(\frac{V_1}{V_2}\right)^{n-1} \right] \frac{V_1}{m_n} \qquad (8)$$

For nitroglycerine powders, such as the rifle cartridge powder of the "Sokol" mark used in explosion units

$$v_p \approx 3.16.10^3 \sqrt{\frac{\left[1 - \left(\frac{V_1}{V_2}\right)^{0.2} \right] V_1\Delta}{(1-0.8\Delta)\, m_n}} \quad dm/sec \qquad (9)$$

Varying the weight of the powder charge, the mass of the plunger and the length of the barrel, we can arrive at a value of the design velocity equal to the optimum velocity.   In doing so the weight of the powder must be increased by 8 to 10% of the calculated weight.   This will compensate for any incomplete combustion in the powder.

It should be noted that for the majority of holes the diameter of the part of the plunger sliding along the barrel (with a sliding fit of $\frac{A_3}{C_3}$ or $\frac{A_3}{X_3}$) is 1.5 to 2 times greater than the diameter of the maximum hole punched by the projected unit.   The choice of this diameter is based on the provision of an adequate bearing surface when the plunger approaches the detail in question.

From the calculations of the approach velocity of the punch, therefore, a choice can be made of a suitable barrel for the unit.   Assuming a given type of cartridge containing the necessary powder charge, the cartridge chamber and the lock of the unit can be designed.   The chamber and lock can be of the rifle type with a breech, or of the pistol type.

The dimensions of the bracket or frame, that is the distance between the ends of its two arms (one of which is threaded for adjusting the position of the barrel and the other contains the die) and the radius of curvature of the frame, are determined by the governing conditions ensuring the necessary correct approach to the surface being punched.

From the displacement velocity of the plunger and the maximum pressure of the gases in the barrel, calculated to formulae (9) and (5) respectively, a check calculation can be made for the strength of the main components and sections of the unit, including the frame and the barrel.

The check calculation for the strength of the bracket frame can be made in the following manner.   The frame is regarded as constituting a bent beam, bent along the arc of a circumference of radius R_o (fig.4).   The weight of a unit length of the beam is q.   The weight of the impact block Q is concentrated at point C.   The load P (plunger) travels along the axis y and hits the impact block, which is rigidly fixed to the bent beam at point C.   The

impact block constitutes a massive block in which the die matrix is fixed. This block is assumed to be incapable of any deformation when subjected to impact. The frame (bent beam) is rigidly fixed at point A. The load P travels with a velocity v_o prior to impact.

The equation of the elasticity curve, the curves for the bending moments and transverse forces and the value of the displacement of any point in the bent beam $(f = \sqrt{x^2 + y^2})$, are determined in accordance to the method described in the work of S.D. Ponomarev and others (8).

Since the force P is applied to the frame at point 1 with a velocity v_o it is necessary to introduce a dynamic coefficient in order to design a frame subjected to a dynamic load effect. The dynamic coefficient is determined in a similar manner as for a beam or girder subjected to a shock load:

$$K_\partial = \sqrt{ \frac{v_o^2}{g f_{stat}} \cdot \frac{1 + \frac{Q}{P} + 0.82 \frac{q R_o}{P}}{\left(1 + \frac{Q}{P} + 1.38 \frac{q R_o}{P}\right)^2} } \qquad (10)$$

Knowing this coefficient, it is not difficult to determine the stresses and displacements at any point of the beam when subjected to the effect of a shock load:

$$\sigma_{dyn} = K_\partial \sigma_{stat} \qquad (11)$$

$$f_{dyn} = K_\partial f_{stat} \qquad (12)$$

where σ_{stat} and f_{stat} are the stress and the displacement respectively due to a statically applied load.

Fig.4   Calculation diagram for the frame.

As is well known the most dangerous section of the bent beam (frame) is the section at point 5 (fig.4), which is situated opposite the point at which the force is applied. This is the section which must be checked for strength.

With static loading the bending moment at this section is

$$M_{calc} = M_{max} = - 2PR_o \qquad (13)$$

At the same section the tensile force is

$$N_{calc} = N_{max} = P \qquad (14)$$

Since the dynamic coefficient can only be calculated for an actual existing frame, we must in the first place assume preliminary given dimensions of the frame and then check the section of this given frame for strength, allowing for the shock effect of the load.

To simplify the calculations, the sectional dimensions of the frame can be expressed in terms of its main overall parameters (fig.5) as follows:

$$R_1 = h = 4h^1 = 4b; \quad h^1 = b; \quad b_2 = b;$$
$$R_2 = R_1 + h^1 = 5b; \quad b_1 = 3b;$$

5.

$$R_3 = R_1 + h = 8b; \quad h'' = 3b.$$

The distance to the centre of gravity of the section will then be

$$R_0 = R_1 + 1.5b = 5.5b.$$

The area of the frame section $F = 6b^2$ and the radius of the neutral layer is

$$r = \frac{F}{b_1 \ln \frac{R_2}{R_1} + b_2 \ln \frac{R_3}{R_2}} = 5.268 \, b. \tag{15}$$

The eccentricity can be determined from the formula

$$Z_0 = R_0 - r = 5.5b - 5.268b = 0.232b$$

and the distance from the neutral layer to the inner fibres is

$$h_1 = R_1 - r = 4b - 5.268b = -1.268b$$

and to the outer fibres it is

$$h_2 = R_3 - r = 8b - 5.268b = 2.732b.$$

The value of the static moment is

$$S = Fz_0 = 6b^2 . 0.232b = 1.392b^3$$

The total stress in the inner fibres is then

$$\sigma_{inner} = \frac{M_{max} h_1}{SR_1} + \frac{N_{max}}{F} \tag{16}$$

and in the outer fibres it is

$$\sigma_{outer} = \frac{M_{max} h_2}{SR_3} + \frac{N_{max}}{F} . \tag{17}$$

Substituting the values of all the terms contained in formulae (16) and (17), we get the values of the stresses in the inner and outer fibres due to a static load:

$$\sigma_{inner} = \frac{2.7P}{b^2} \tag{18}$$

$$\sigma_{outer} = \frac{2.53P}{b^2} \tag{19}$$

The stresses due to a dynamically applied force can be determined from formulae (10) and (11).

If we assume very approximately that $Q=3P$ and $qR_o=2P$, then from formula (10)

$$K_\partial = \frac{v_o \cdot 0.35}{\sqrt{gf_{stat}}} \tag{20}$$

$$f_{stat} = 8.8 \frac{PR_o^3}{EJ'} = \frac{263P}{Eb} \tag{21}$$

where J' is the approximated moment

$$J' = SR_1 = 1.39b^3 \cdot 4b = 5.56b^4$$

The condition of stability for the frame is

$$\sigma_{dyn} < \sigma_{stat} \tag{22}$$

or

$$\frac{2.7P}{b^2} \cdot \frac{0.35 v_o}{\sqrt{gf_{stat}}} < \sigma_{stat} \tag{23}$$

Substituting in this last expression the value for f_{stat}, we get

$$\frac{3Pv_o^2 E}{10^3 gb^3} < \sigma_{stat}^2$$

whence

$$b \geq \sqrt[3]{\frac{3Pv_o^2 E}{10^3 g\sigma_{stat}^2}} \tag{24}$$

This formula enables us to determine the approximate thickness of the T-section wall from conditions of stability.

After calculating the thickness to this formula, we re-calculate all the dimensions of the frame to the given relationships above. If we find that $4b < R_1^*$ (where R_1^* is the constructive dimension of the frame chosen on the basis of its operational duty), then b must be taken as equal to $\frac{R_1^*}{4}$ and then from this parameter all the other dimensions of the frame can be deduced.

Another important element in the construction of the explosion punching unit is the barrel, the section of which is checked from results of ballistic calculations of the pressure in the channel of the barrel. The check calculation for the barrel section is carried out in accordance with the method of A.V. Gadolin, applicable to thick-gauge cylinders (9). Taking the inside pressure to be equal to p_1 as given by equation (5) and the outside pressure to be equal to zero, we can deduce the wall thickness of the barrel of the projected unit. We see, therefore, that the check calculation for strength enables us to finalise the dimensions of the barrel and determine the basic dimensional parameters of the frame.

Silencing during operation. One of the chief conditions for a wider application of the explosive method of hole punching in industry is the elimination of sound during the process.

To solve this problem (even if only partially), it was decided to measure the level of the sound pulses arising in the process of punching the holes, both with and without the use of silencers of different construction (fig.6). A special apparatus based on the design of the mark 'ShI-1' noise gauge (registered patent No.23631 in the names of V.G. Kononenko, K.I.Zaitsev and D.A. Raizman), incorporating radio-electronic elements, was used for measuring the sound characteristics. The results of the measurements have shown that the most successful, both from the point of view of sound reduction and design, is the silencer shown in fig.6 b. In this type of silencer the silencing is effected in the same way as in the usual silencers of the labyrinth or baffle type (fig.6 c).

Fig.6 Constructional diagrams for silencers.

a - first type of silencer: 1- barrel of gun; 2- body of silencer; 3- perforated lining; 4- rubber; 5- clamping rings; b- second type of silencer: 1- barrel of gun; 2- frame; 3- silencer casing; c- labyrinth system of silencer: 1- connecting piece from the exhaust port; 2- body of silencer; 3- labyrinths; 4- exhaust nozzle.

The gases, escaping through the exhaust ports in the barrel, impinge in the first place against the inner recess cavity of the frame and then, after doing a certain amount of work, escape through the openings in the frame and encounter in their path the casing of the silencer, which is swaged outwards. Here again a certain amount of energy is absorbed by the work done. The pressure of the exhaust gases is thus considerably reduced, approaching atmospheric pressure. The process of punching the hole is thus almost silent and all that is heard is the sound of the impact.

A general view of a typical unit of the explosion device is shown in Fig.7.

Fig.7 General view of a typical unit for punching holes by explosion

Choice of dimensions and shape for the cutting edges of the tool. The shape and dimensions of the tool used, as well as its heat treatment, must be chosen in such a way, as to ensure not only a clean and accurate hole, but also an easy withdrawal of the plunger from the part. An adequate stability of the tool must also be ensured.

Dies and plungers can be made of mark 'U8A' tool steel, heat treated to a hardness of RC52-56. To ensure stable conditions, the diameter of the bearing surface of the die piece is made about 3 to 3.5 times larger than the diameter of the working portion of the die surface.

The height of the cutting edge of the die can be 3 to 4 mm (this is based on conditions ensuring stability and withstanding repeated grinding).

In order to ensure a free withdrawal of the plunger from the part, it is recommended to use plungers with a conical working surface (angle of cone 3 to 5°).

The height of the working surface of the plunger is based on ensuring adequate conditions for punching the hole and for forcing the cut material through the hole and die.

Tests have shown that the use of tools of recommended dimensions for explosion punching yields good, cleanly finished holes of adequate accuracy. The stability of the tool with high velocities of the plunger, which obtain in explosive punching, is considerably higher than that of the usual hole punching stamping dies.

The above data have been tested and approved in experimental projects and in working industrial installations.

Bibliography

1. V.G. Kononenko and K.I. Zaitsev. Noiseless explosive punching of holes. "Kuznechno-shtampovochnoe proizvodstvo" ("Forging and stamping industry"), (12),1959.

2. K.I.Zaitsev. Explosive punching of holes in the manufacture of railway trucks. "Izvestiya vuzov. Mashinostroenie" ("Universities Bulletin. Machine Construction"), (4), 1961.

3. V.G. Kononenko. Explosion impact machine for high speed tests. "Zavodskaya laboratoriya" ("Works Laboratory"), (3), 1959.

4. M.B. Zubtsov. Sheet metal stamping. Mashgiz, 1958.

5. S.P. Gubkin. Treatment of metals by pressure. Metallurgizdat, 1947.

6. A.S. Yastrezhembskii. Technical thermodynamics. Gosenergoizdat, 1960.

7. M.E. Serebryakov. Internal ballistics. Oborongiz, 1949.

8. S.D. Ponomarev and others. Strength calculations in machine construction, 3, Mashgiz, 1959.

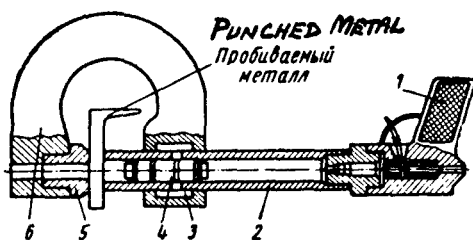9. N.M. Belyaev. Strength of materials. Gostekhizdat, 1954.

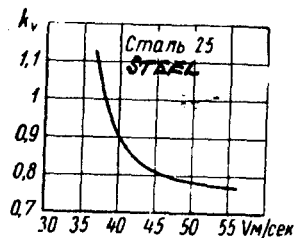Рис. 1. Схема установки для взрывной пробивки отверстий.

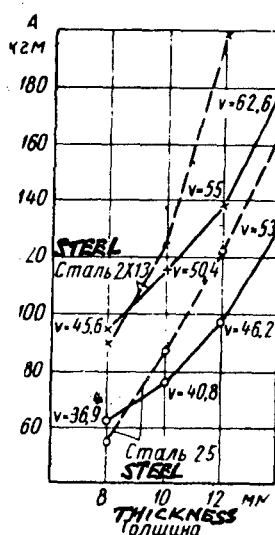Рис. 2. Зависимость коэффициента динамичности от скорости подлета пуансона.

Рис. 3. График зависимости работы деформирования от толщины пробиваемого материала и скорости подлета пуансона:
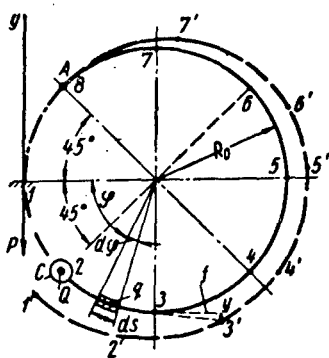
————— взрывная пробивка;
— — — пробивка на прессе.

Рис. 4. Расчетная схема скобы.

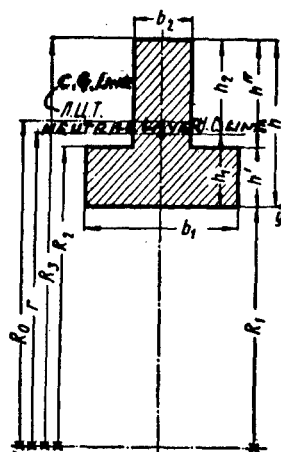Рис. 5. Сечение скобы.

Рис. 7. Общий вид типовой установки для взрывной пробивки отверстий.
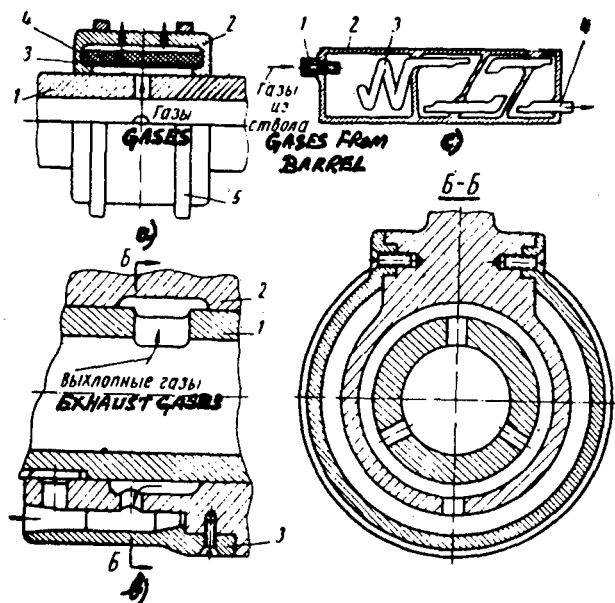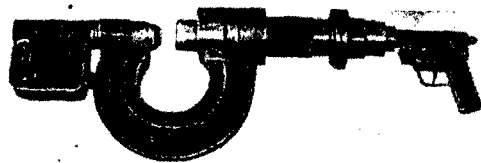

Рис. 6. Конструктивные схемы глушителей:

а — первый вариант глушителя; 1 — ствол установки; 2 — корпус глушителя; 3' — перфорированная прокладка; 4 — резина; 5 — стяжные кольца; б — второй вариант глушителя; 1 — ствол установки; 2 — скоба; 3 — кожух глушителя; в — лабиринтная система глушителя звука; 1 — штуцер выхлопного окна; 2 — корпус глушителя; 3 — лабиринты; 4 — выхлопной патрубок.

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